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AERODYNAMIC AND THERMAL DESIGN OF CENTRIFUGAL COMPRESSOR FOR SMALL SCALE IN NON-DIMEMSIONAL FORM

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Original Article

ABSTRACT

In the present study, a 3D numerical simulation of a non-dimensional design of impeller, diffuser and volute design of centrifugal compressor in dimensional form is carried out by using the commercial code of ANSYS R15.0. The fully 3D, Compressible and Steady flow analysis of the fluid (air) flow were solved in CFX solver. A SST k-v turbulent model was used for a simulation of the on-design performance of the single speed centrifugal compressor. By using this specific design results show that De-Haller number, non-dimensional mass flow rate, non-dimensional ratio of axial length to exit diameter, divergence angle of Diffuser, area ratio from diffuser outlet to volute outlet obtained in order to 0.7, 0.092, 0.33, 7.29 and 0.65 respectively. Also, found that All Mach number are below 1 so there are no shock formations. The mass flow rate is less than chocking mass flow rate at the throat of impeller and diffuser so there is no choking of flow In order to verify the present numerical simulations, the model results were validated with the analytical method gives the good agreement but not exactly due to design losses were neglected.

KEYWORDS: Non Dimensional, Centrifugal Compressor, De-Haller Number, Non-Dimensional Mass Flow Rate, Mach Number, SST k-v Turbulent Model & Aero Dynamic Flow Etc

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INTRODUCTION

Centrifugal compressor, also called radial compressor, are critical equipment in a wide variety of application in the chemical process industries, power plant etc. As their name suggest, their primary process is to compress a fluid in to smaller volume while simultaneous increasing pressure and temperature of fluid. In other words, compressor accepts a mass of gas at some initial pressure and temperature and rise pressure and temperature of a gas.

Working Principle

The flow enters a three dimensional impeller through an accelerating nozzle and a row of inlet guide vanes as shown in fig 1. A centrifugal compressor consists of a rotating impeller followed by diffuser. Fluid is drawn in through the inlet casing into the eye of the impeller. The function of the impeller is to increase the energy level of the fluid by whirling outwards, thereby increasing angular of the fluid. Both the static pressure and the velocity are increased within the impeller. Static pressure rise due to relative flow deceleration as well centrifugal force action. This process can be accomplished by free diffusion in the annular space surrounding the impeller or by incorporating a row of fixed diffuser vanes, which allows the diffuser to be made very much smaller. Outside

the diffuser is a scroll or volute whose function is to collect the flow from the diffuser and deliver it to the outside pipe.

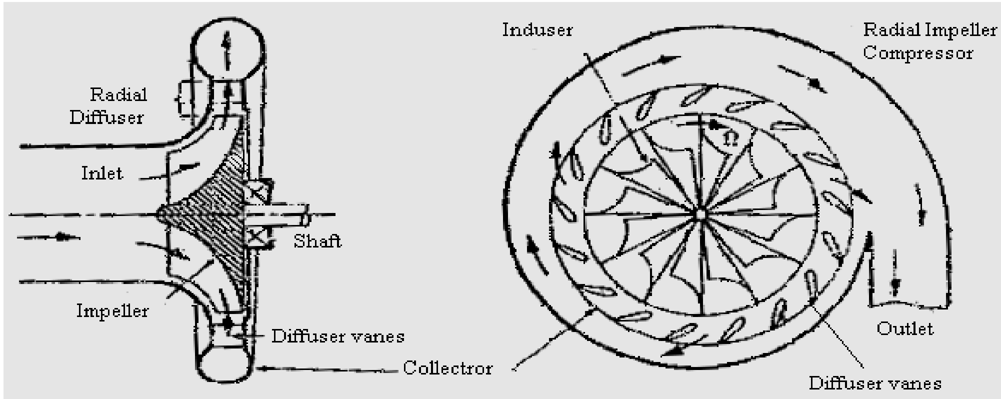

Figure 1: Typical Radial Compressor

Non-Dimensional Design of Centrifugal Compressor

The user of turbo-machines will generally require parameters which readily describe the overall performance of the machine so that assessment and comparison can be easily made. The designer requires parameters which will enable him to select the correct machine and make valid comparison between competing designs. In addition to this the designer will requires parameters to assess the detailed performance of individual components of the machine so that deficiencies can be easily identified, and design improvements implemented.

A comprehensive literature review of centrifugal Compressor has been involved in design and performance prediction, analysis and flow investigation, experimentation of key components as impeller, vane less and vane diffuser, volute casing etc. has been provided in references [1-33].*Hua Chen &Vai-Man Lei (2012)* [2] found that ported shroud is a cost-effective casing treatment that can greatly improve stability of centrifugal compressors. *Abraham Engeda, Yunbae Kim (2003)* [3]worked on the inlet flow structure of a centrifugal compressor stage and its influence on the compressor performance. The performance of centrifugal compressor can be seriously degraded by inlet flow distortions that result from an unsatisfactory inlet configuration.*Barend W. Botha and Adriaan Moolman (2005)* [4] generated idea that by use of computational performance prediction methods has led to significant improvement turbo-machine technology. *PekkaRoytta et.al. (2009)*[5]Performed a hypothesis to design an impeller flow passage for a centrifugal compressor is tested. *Miles coppinger (1999)*[6]worked on aerodynamic performance of an industrial centrifugal compressor variable inlet guide vanes systems. Industrial centrifugal air compressor can require a combination of large range of mass flow, high efficiency, constant pressure ratio and constant rotational speed. *Neil paul Murray (2003)*[7]performed an experiment on effect of impeller diffuser interaction on centrifugal compressor performance. The effect of impeller diffuser interaction in a centrifugal compressor on the performance of the stage was examined using unsteady 3-dimenstinal Reynolds average Navier-Stokes simulations.*James M. Sorokes et al. (2000)*[8] performed investigation of the circumferential static pressure non-uniformity caused by a centrifugal compressor discharge volute. They described experimental and computational fluid dynamics analyses of the non-uniform static pressure distortion caused by the discharge volute in high pressure, centrifugal compressor using a heavily instrumented gas re-injection compressor operating at over 6000 psia discharge.*Adnan HamzaZahed and NazihNoamanBayomi(2014)*[9] investigated the development of a preliminary design method for centrifugal compressor. The design process starts with the aerodynamic analysis of the preliminary design and its reliance

on empirical rules limiting the main design parameters. *C.Xu and R.S.Amano (2012)*[10] worked on meridional considerations of the centrifugal compressor development. They found that high pressure ratio and efficiency of the centrifugal compressor require impeller design to pay attention to both the blade angle distribution and meridional profile. The geometry of the blades and the meridional profiles are very important contributions of compressor performance and structure reliability. Proper meridional profiles can improve the compressor efficiency and increase the overall pressure ratio at the same blade back curvature. They concluded from this study, a proper design for the inlet and the exit of the meridional plane can improve the compressor overall efficiency and also can reduce the sensitivity of the tip clearance. *FloreCrevel et al. (2014)*[11] worked on concept of deep surge. Aerodynamic instabilities such as stall and surge may occur in compressors, possibly leading to mechanical failure so their avoidance is crucial. Aerodynamic instabilities in compressors that occur at low mass flow, such as stall& surge, can induce a total breakdown of the flow and very damaging for the machine. *FloreCrevel et al. (2014)*[12]also worked on Rotating stall, The influence of the volumes surrounding the compressor on rotating stall flow pattern they found evidence that the ratio between the volume of the compressor and downstream volumes has an impact on unstable frequencies. *D pan et al. (1999)*[13] suggested initial conceptual design of centrifugal fan and compressor volutes and extended to accommodate overhung volute designs often used in process and turbocharger compressors. The numerical computational results also showed that by enlarging the flow passage area under the tongue the internal flow structure of the volute was improved at off-design flow conditions and at low flow rates the flow under the tongue became more uniform and at large flow the size of the separation region in the exit duct was reduced.*I Bennet et al. (2000)*[14] claimed that pipe diffusers, as potentially being an improvement over conventional vane island diffusers improving compressor efficiency up to 2-3 per cent, particularly at higher pressure ratios. They also suggested that geometry of the region between impeller tip and diffuser throat is accepted as being very influential, crucial to the performance of the centrifugal compressors. *G. Naga MalleshwarRao et al. (2014)*[15] estimated the impeller geometry from pressure ratio and mass flow rate of the refrigerants are arrived from the thermodynamic cycle analysis of refrigeration cycle. *Dr. Basharat Salim(2013)*[16] estimated effect of geometrical parameters on the performance of wide angle 5^0, 7^0,10^0 and12^0 were used to find the effect of diffuser angle, where as another four diffusers with area ratio 1.56, 1.76, 1.97and 2.24 were used to determine effect of area enlargement for a diffuser angle of 7^0. The Reynolds numbers at which investigations were carried out were calculated using free upstream velocity which was measured upstream of the diffuser inlet so as to avoid the influence of the diffuser on velocity profile. Also the diffuser with higher area ratio show higher values of pressure recovery compared to the lower area ratio diffusers. The diffuser efficiency of diffuser with diffusing angle of 7^0 is found higher than all the other diffusers at all the Reynolds numbers. Diffuser efficiency seems to be more influenced by the Reynolds number than the area ratio of these diffusers.*Moroz (2010)*[17] investigated the effect of changing the stagger angle of various vane diffuser designs. He performed tests on three diffusers by changing the stagger angle of each. When increasing the stagger angle by 3 degrees, a slight increase in pressure recovery is observed, at the expense of flow rate, whereas by decreasing the stagger angle by 3 degrees, the flow rate is increased with a slight decrease in pressure.*Xi et al. (2007)*[18] found that by reducing the diffuser stagger angle, i.e. smaller than the design value, the maximum pressure ratio increases at the expense of compressor efficiency and flow rate. *Loringer et al. (2006)*[19] investigated the effect of boundary layer growth on the pressure and suction walls of the diffuser and suggested flow slots in the diffuser vanes. They claimed the flow slots in the diffuser vanes will minimize the growth of a flow separation zone along the suction side. *Rayan et al. (1980)*[20] investigated the effect of high swirl on vane diffusers. They showed that the length of the vaneless space between the impeller exit and diffuser leading edge is a major factor in diffuser efficiency. Finally, they achieved a minimum loss coefficient when the vane leading edge radius is

approximately 1.2 times the vaneless diffuser inlet radius,R3/R2= 1.2. *Aungier (2000)*[21] recommended a vane-less radius ratio(r3/r2) of between 1.06 and 1.12, i.e.1.06<r3/r2<1.12. The lower limit of 1.06 is used to provide space for the distorted impeller flow to smooth out and the blade wakes to decay before the flow enters the diffuser vanes. The upper limit of 1.12 is used due to the low-flow angles involved that occurs as a result of the high impeller tip Mach numbers, resulting in high vane-less space loss levels. *A.Whitfield and M A. Johnson (2002)*[22] demonstrated, A satisfactory fabrication procedure for volutes and it has been shown that the use of a square section did not have a significant adverse effect on volute performance. *Ahmed S. Hassan (2006)*[23] worked on influence of the volute design parameters on the a centrifugal compressor of aircraft turbocharger range of stable operation and pressure rise coefficient as well as diffuser pressure recovery factor has been investigated theoretically and experimentally.

The present new study focuses on develop a non-dimensional aerodynamic design methodology of centrifugal compressor Impeller. To develop Vane and Vane less Diffuser and volute design methodology in dimensional form. Also, Numerical simulation of proposed design.

THEORETICAL MODEL FOR WHOLE CENTRIFUGAL COMPRESSOR

Conceptual Design of Centrifugal Compressor Impeller

The main requirement from an impeller design procedure is the computation of the overall principal dimensions and the inlet and discharge blade angles. Impeller design procedure is carried out applying non dimensional parameters thermodynamic correlation which disregard actual size of machine and more general compared to dimensional quantities. Impeller design has been accomplished systematically for complete control of aerodynamic parameters within optimum recommended range. The preliminary dimension of centrifugal compressor impeller and calculated resultsfrom the various correlations and equations i.e. Stodala, Weisner, Stanitz, Lame Ovals, equations&Birdi correlations, Johnston and Dean (1966), Rodger and Sapiro (1972) equationare shown in followingtable 1& 2

Table 1: Preliminary Data of Centrifugal Compressor Impeller

P	**25000 W**	α_2	**70^0**
PR	**3**	β_{B2}	**0**
T01	**298 K**	α_1	**0**
D1h/D1s	**0.35**	μ	**0.85**
D1s/D2s	**0.55**	C_p	**1005 J/kg*K**
η_i	**0.9**	R	**287 j/kg*K**
η_s	**0.8**	Υ	**1.4**

Table 2: Solution of Preliminary Data of Centrifugal Compressor Impeller

λ	**0.85**	P_{02}/P_2	**1.83**
U2/a01	**1.164**	P_2/P_{01}	**1.84**
T_{02}/T_{01}	**1.46**	ρ_1/ρ_{01}	**0.933**
P_{02}/P_{01}	**3.368**	ϕ	**0.078**
$C'_{02}/a01$	**0.989**	θ	**0.092**
C_{m2}/a_{01}	**0.423**	b_2/R_2	**0.0721**
$\alpha'2$	**66.81^0**	**Ns**	**0.663**
C'_2/a_{01}	**1.076**	**m**	**0.1811 kg/s**
C'_2/a_{02}	**0.89**	ρ_{01}	**1.1847 kg/m3**
T_2/T_{02}	**0.821**	a_{01}	**346.029 m/s**
M2	**0.971**	A_2	**0.005 m2**
U2/a02	**0.963**	R_2	**0.039 m**

Table 2: Contd.,			
U2/a2	1.05	b_2	0.003 m
C'θ2/a2	0.892	R_{1s}	0.0215 m
$W'_{\theta 2}/a_2$	-0.157	R_{1h}	0.0075 m
M2'	0.413	D_{1rms}	0.0345 m
β_2	-22.39	U_{1rms}	166 m/s
U_1/a_{01}	0.64	W_{1rms}	210 m/s
$W_{\theta 1}/a_{01}$	-0.64	m_c	0.204 kg/s
C_1/a_{01}	0.369	DH	0.7
M_1	0.374	Z_b	20
M_1'	0.749	U_2	402.89 m/s
M_R	0.551	N	98177 rpm

Conceptual Design of Centrifugal Compressor Diffuser

Centrifugal compressors and pumps are, in general, fitted with either a vaneless or a vaned diffuser to transform the kinetic energy at impeller output into static pressure. The diffuser system of a centrifugal compressor usually constructed of either vaneless or a combination of a vaneless diffuser followed by a vaned diffuser. The vaneless diffuser is often adopted as sole means of pressure recovery owing to its simplicity and inexpensive construction. The simplest description of the flow through the vaneless diffuser can be obtained by considering the angular momentum equations. For the case of open passage the flow is only restrained by the sidewalls, and in the absence of any wall friction force, the torque exerted on the fluid is zero and the angular momentum reduces to the free vortex relationship.

C_θ: R = constant

By combining with the continuity equation it can be shown that the fluid flow angle through vaneless space is given by:

$$\frac{\rho b}{\tan\alpha} = \text{const.}$$

For a parallel-walled radial vaneless diffuser with assumption of incompressible flow it follows that the flow angle remain constant and the path describes a logarithmic spiral.

$\tan\alpha = C$ when ρ and b are constant

The preliminary dimension of centrifugal compressor diffuser and calculated results from the various correlations and equations i.e. Rodgers (1982), Came and Herbert (1980) and Osborne al. (1975) equations are shown in following table 3 & 4.

Table 3: Preliminary Data of Centrifugal Compressor Diffuser

R3/R2	1.1	R4/R3	1.3
θ_{dv}	0	Σ	1

Table 4: Solution of Preliminary Data of Centrifugal Compressor Diffuser

Ar	1.1	θ_d	7.29^0
$C_{\theta 3}$	311.32 m/s	AR	1.819
T_3	380.18 K	C_{pid}	0.698
P_3	2.12 bar	K_d	0.168
M_3	0.85	P_4	2.726 bar
ρ_3	1.972 kg/m^3	P_{04}	3.19 bar

Table 4: Contd.,			
C_{r3}	119.85 m/s	T_4	419.21 K
α_3	69^0	ρ_4	2.326 kg/m^3
Φ	10^0	A_4	0.001 m^2
AOA	7^0	C_4	156 m/s
C	0.598 m	C_{r4}	78.16 m/s
C_r	0.029 m	$C_{\theta 4}$	135 m/s
Z_v	13	M_4	0.38
I	2	ξ_d	83.1 %
α_{3b}	67^0	A_{3n}/A_{3th}	0.977
α_{4b}	57^0	m_{cd}	0.19 kg/s
α_4	60^0		

Conceptual Design of Centrifugal Compressor Volute Casing

The spiral-shaped volute or scroll casing collects and guide the flow from diffuser or impeller and finally discharges it through outlet delivery pipe. Volute may be of symmetrical or overhung type. The volute is usually designed through application of a one dimensional analysis assuming a constant angular momentum (free vortex) from the volute inlet to the centre of the volute passage. The design objective is to achieve uniform flow distribution without any pressure distortion around circumference of volute passage. The preliminary dimension of centrifugal compressor volute casing and calculated results from the various correlations and equations are shown in following table 5 & 6.

Table 5: Preliminary Data of Centrifugal Compressor Volute Casing

R_{4s}/R_4	1.005	b_4	0.003 m

Table 6: Solution of Preliminary Data of Centrifugal Compressor Volute Casing

$C_{\theta 4s}$	134.32 m/s	Q	0.078 m^3/s
C_{r4s}	77.77 m/s	M_5	0.23
T_{4s}	423.35 K	P_5	3.044 bar
M_{4s}	0.376	θ_t	7.7^0
P_{4s}	2.897 bar	R_t	0.0605 m
ρ_{4a}	2.326 kg/m^3	R_{5v}	0.101 m

RESULTS AND DISCUSSIONS

The numerical modeling & simulation of the on-design performance of the centrifugal compressor stage was considered for single design speed, 3D impeller with 20 twisted blades,

Vane Diffuser with 13 De swirl vanes was successfully carried out. Analysis of flow in a 3D Impeller is done for Single mass flow rate. The contour plots are generated for better understanding of fluid flow through centrifugal compressor. The results obtained from CFD analysis were validated with the theoretical results for performance parameters such as total pressure ratio, static pressure, static temperature Mach number and velocity of a centrifugal compressor stage as estimated by CFD tools almost complies, with variation of 1-9 %, with theoretical results.

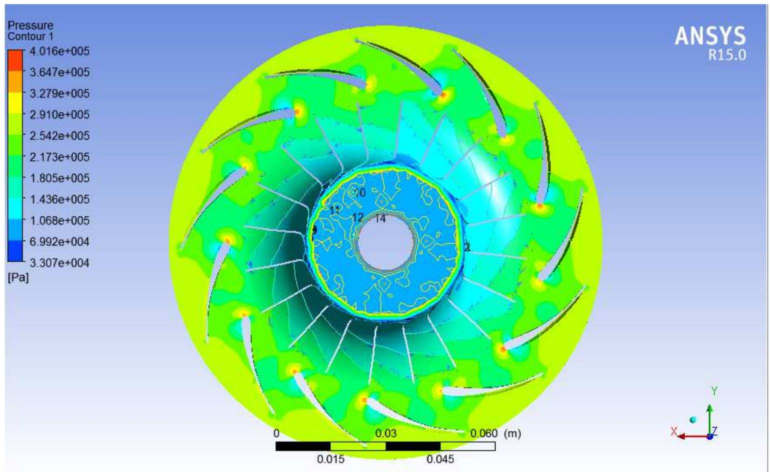

Figure 2: Static Pressure Distribution

Figure 2 ANSYS graphical representation showed the static pressure difference across a centrifugal compressor, it is clearly seen that static pressure is increasing. Its value is good agreements with variation just 1-2 %. Also, observed that pressure increase from 0.9 to 1.9 bar and from 1.9 to 2.8 bar in stationary domain.

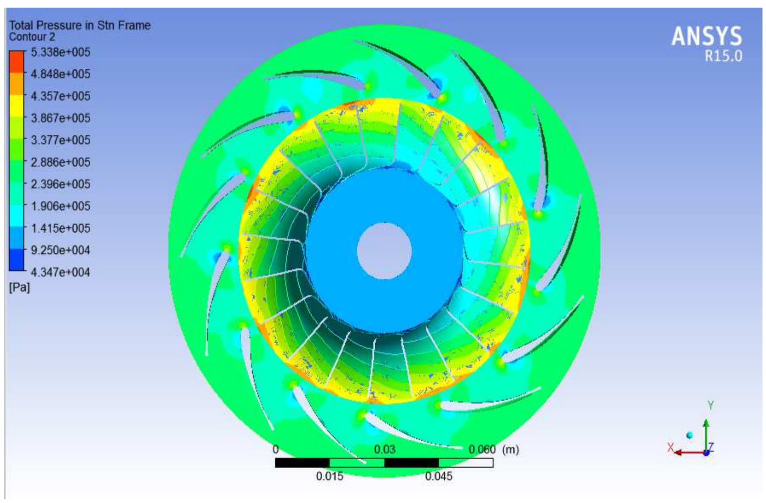

Figure3: Total Pressure Distribution

In Figure 3, It was observed from the graph which showedthe total pressure difference across a centrifugal compressor, it is clearly seen that total pressure is increasing in rotating frame and some pressure loss in stationary frame. Variation in results obtained from CFD solver are 2-3% from theoretical calculation. Also, seen that total pressure increased from 1.01325 to 3.6 bar and again it reduced from 3.6 to 2.9 bar due to losses.

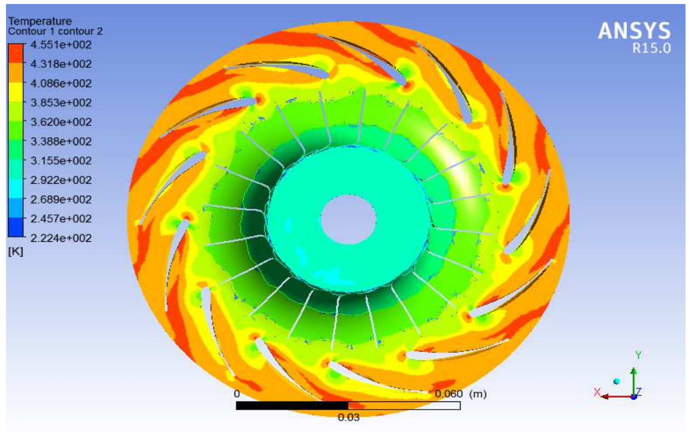

Figure 4: Static Temperature Distribution

It was observed that from the CFD static temperature distribution resultsshowed the static temperature difference across a centrifugal compressor, it is clearly seen that static temperature is increasing in rotating frame and in stationary frame. Variation in result obtains from CFD are 2.5-3.5 % different from theoretical calculation.

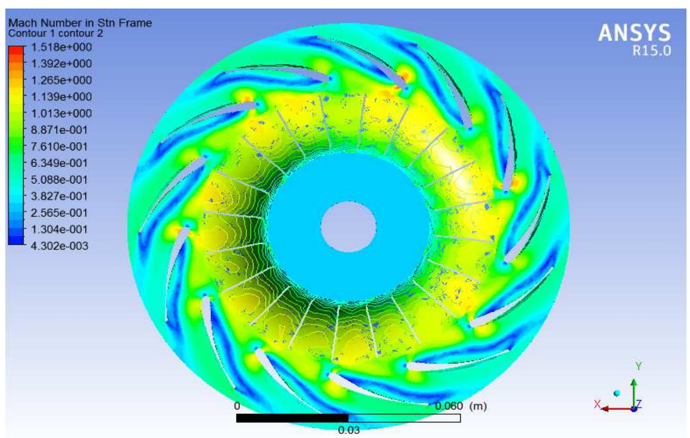

Figure 5: Mach Number Distribution

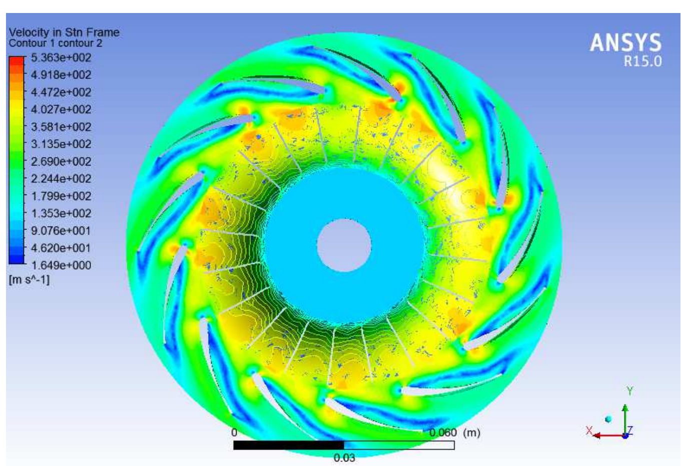

Figure 6: Absolute Velocity Distribution

Figure 5 shows the Mach number difference across a centrifugal compressor it is clearly seen that it is increasing in rotating frame and decrease in stationary frame. Also, found that variation in result obtained from CFD was 5-7 % from theoretical calculation method. Figure 6 shows the Velocity difference across a centrifugal compressor it is clearly seen that it is increasing in rotating frame and decrease in stationary frame. Also, concluded that Variation in result obtains from CFD 8-9 % different from theoretical calculation method.

CONCLUSIONS

In the present study, 3D numerical simulations are conducted to investigate the non-dimensional aerodynamic design of centrifugal compressor impeller, diffuser and volute design in dimensional form. The main conclusions from this specific design are summarizedas follows:

- De-Haller number & Non-dimensional mass flow rate obtained 0.7& 0.092 respectively which was in the range mentioned in literature survey [*Adnan Hamza Zahed et al.2014*]
- Non-dimensional ratio of axial length to exit diameter is 0.33 which was also in the range suggested by Birdi.
- All Mach number obtained below 1 so there are no shock formations.
- Mass flow rate is less than chocking mass flow rate at throat of impeller and diffuser. Therefore, there is no choking of flow.
- Divergence angle of diffuser obtain by this specific design is 7.29 which was in the range mentioned in literature survey [*Dr. Basharat Salim.et al.2013*]
- Area ratio from diffuser outlet to volute outlet is 0.65 which was in the range mentioned in literature survey [*Ahmed S. Hassan et al.2006*]
- Results obtained from the CFD solver and theoretical calculation method are good agreement with each other but not exactly, due to neglected the design losses.

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APPENDICES

Nomenclature

Table 1: Symbols

A	Flow area section of impeller, diffuser & volute channels
a	Speed of sound
AOA	Design angle of attack
C	Absolute velocity of gas (Air)
W	Relative velocity of gas with respect to rotating element
C_p	Specific heat at constant pressure, Static pressure recovery
D	Impeller exit (tip) diameter
i	Blade incidence angle
M_u	Blade Mach number
M, M'	Absolute & Relative Mach number
m	Mass flow rate
m_c	Chocking mass flow rate
N	Rotational speed of Impeller
n_{ss}	Specific speed
P	Fluid static pressure
PR	Stage total pressure ratio
P_o	Fluid Stagnation pressure
R	Universal gas constant, Radius
T	Temperature
t_b	Impeller blade thickness
U_2	Blade tip speed
W	Euler work done, Relative velocity
x	Meridional (axial) distance from axis of rotation
Z_b	Number of Impeller blades
Z_v	Number of diffuser vanes
z	Axial meridional distance along axis of rotation
C_d	Chord length of vane diffuser in rectilinear coordinate system
C_r	Chord length of vane diffuser in curvilinear coordinate system
x,y	Coordinate of rectilinear coordinate system
r,θ	Coordinate of curvilinear coordinate system
Q	Flow Rate
r	volute section radius

Table 2: Special Characters

Α	Absolute flow angle w.r.t meridional (flow) direction
β	Relative flow angle w.r.t flow direction
β_B	Blade angle
λ	Work input coefficient
γ	Isentropic index for air, staggered angle in case of diffuser
ν	Impeller eye hub to shroud diameter ratio
μ	Slip factor
θ	Non-dimensional mass flow factor
θ_{TE}	Vane diffuser Trailing edge angle in curvilinear system
θ_d	Vane diffuser divergence angle
σ	Solidity of vane diffuser
η	Efficiency
φ	Camber angle

Table 3: Subscript

01	Stagnation state at inlet to eye
02	Stagnation state at impeller exit
03	Stagnation state at vane-less exit
04	Stagnation at exit of vane diffuser
04s	Stagnation at inlet of volute
05	Stagnation at exit of volute
1	Static state at impeller eye
2	Static state at impeller exit
3	Static state at vane-less exit
4	Static state at exit of vane diffuser
4s	Static state at inlet of volute
5	Static state at exit of volute
1s	State at shroud of impeller eye inlet
1h	State at hub of impeller eye inlet
1m	State at mean section of impeller eye inlet
a	Axial meridional component
c	Chock condition
r	Radial component
m	Meridional component
θ	Tangential component
t	Tongue
th	Throat